Unit 2 Reaction Turbine

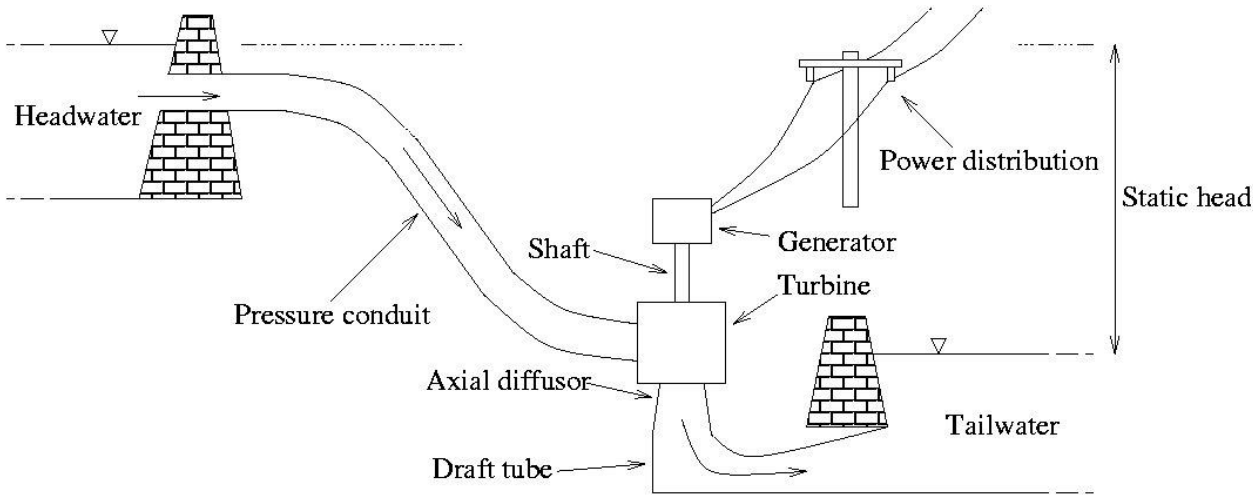
In reaction turbines the runner utilizes both potential and kinetic energies. As the water flows through the stationary parts of the turbine, whole of its pressure energy is not transferred to kinetic energy. When water flows through the moving parts the remaining pressure energy is converted to kinetic energy. Energy conversion from PE to KE is gradual. Inlet pressure is always higher than the outlet pressure, draft tube is required.

Types of Reaction turbine

Most commonly used are Francis and Kaplan turbine.

Francis turbine

It is mixed flow (inwards) type of reaction turbine.



Main parts.

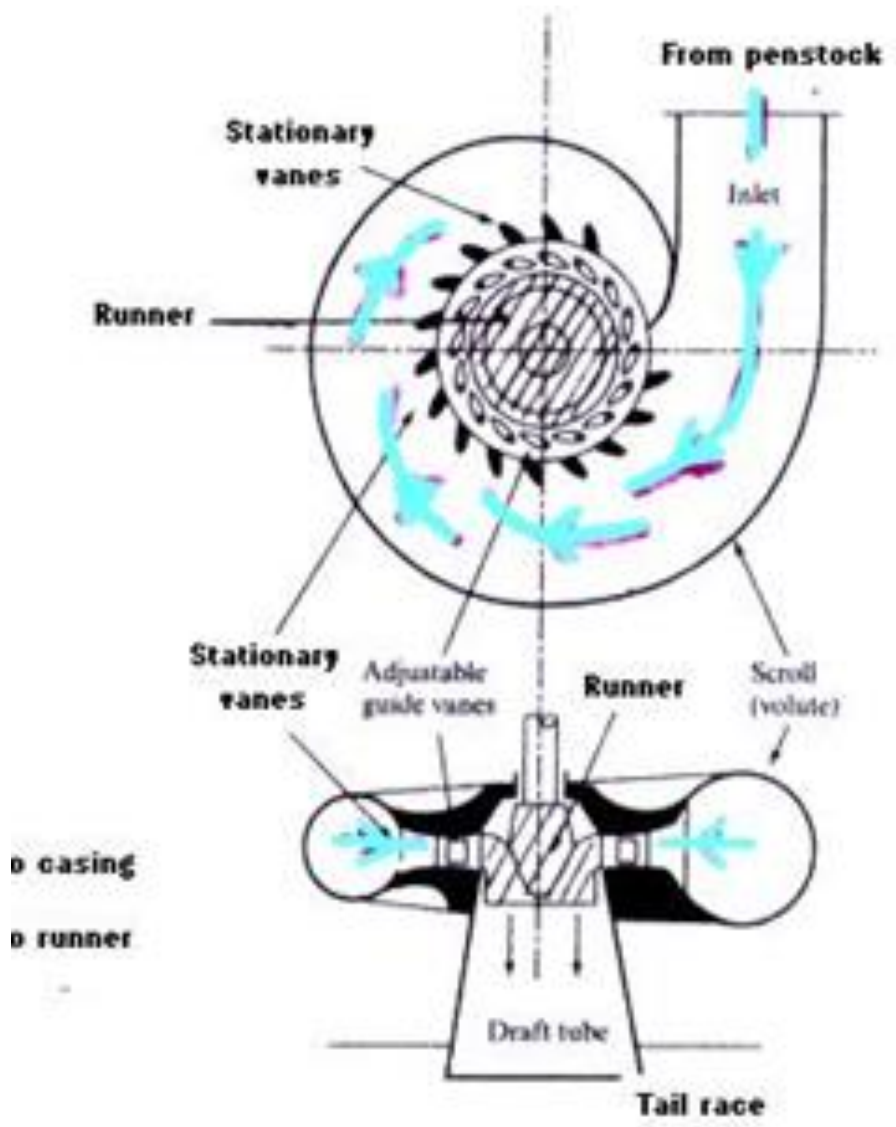
1. Penstock: Large conduit.
2. Spiral/scroll casing: Closed conduit, cross-section area gradually decreases along the flow
3. Guide vanes/Wicket vanes: Directs the water on to runner or stay vanes.
4. Runner(Moving vanes): 16 to 14 vanes
5. Draft tube: Gradually expanding tube which discharges water to tail race.

Heads for Reaction turbine

H_g=Gross head

h_f=loss of head in penstock

$H=H_g-h_f$ (net head)



Also,

H=[Total energy available at exit of Penstock]-[Total energy at exit from draft tube]

= $[\frac{P}{w} + \frac{V^2}{2g} + z]_{\text{penstock}}$ - $[\frac{P}{w} + \frac{V^2}{2g} + z]_{\text{draft tube}}$

If draft tube, tail race level is taken as datum level

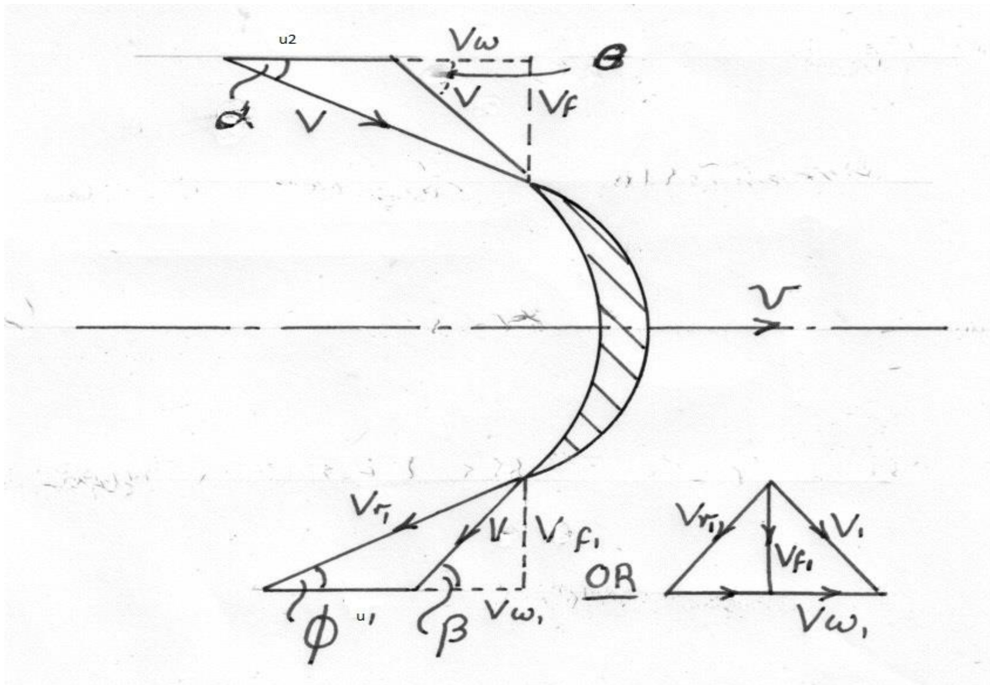
H=$[\frac{P}{w} + \frac{V^2}{2g} + z]_{\text{penstock}} - \frac{V_d^2}{2g}$ Where V_d^2=Velocity at exit of draft tube

W.D. per sec on runner = $\rho a V_1(V_{w1}u_1 \pm V_{w2}u_2)$

= $\rho Q(V_{w1}u_1 \pm V_{w2}u_2)$

Where Q=aV_1=Discharge m^3/sec

W.D. = $\frac{wQ}{\rho}(V_{w1}u_1 \pm V_{w2}u_2)$



Francis turbine

Vanes in series

W.D /sec $=\rho a V_1(V_{w1}u_1+V_{w2}u_2)$

$= \rho Q(V_{w1}u_1+V_{w2}u_2)$

$=\frac{w}{g}\ Q(V_{w1}u_1+V_{w2}u_2)$

$\eta_h = \frac{W.D/sec}{W/P}=\frac{\frac{w}{g}Q(Vw1u1+Vw2u2)}{wQH} = \frac{1}{gH}(V_{w1}u_1+V_{w2}u_2)$

$\eta_{overall}= \eta_{mech}\ \eta_h=\frac{S.P}{R.P}\frac{R.P}{W/P} = \frac{2\pi NT/_{60}}{wQH}$

Energy given to runner = $\frac{wQ}{g}(V_{w1}u_1)$

$H=\frac{V_{w1}u_1}{g} + \frac{V_2^2}{g}$+lost K.E at outlet

Francis Runner

Discharge, Q = $(\pi D_1 B_1)V_{f1} = (\pi D_2 B_2)V_{f2}$

$$\frac{V_{f1}}{V_{f2}} = \frac{D_2 B_2}{D_1 B_1}$$

When Discharge is Radial, (V_2 is radial)

$\beta = 90^o$, $V_{w2} = 0$

$$u_1 = \frac{\pi D_1 N}{60} \qquad u_2 = \frac{\pi D_2 N}{60}$$

$$\therefore \ \frac{u_1}{u_2} = \frac{D_1}{D_2}$$

$$V_{w1} = V_1 \sin\alpha \quad V_{f1} = V_1 \cos\alpha$$

$$V_{r1} = \sqrt{\left(V_{w1} - u_1\right)^2 + V_{f1}^2}$$

$$V_{r2}\cos\phi = u_2 + V_{w2}$$

$$V_2 = \sqrt{V_{w2}^2 + V_{f2}^2}$$

$u = K_u\sqrt{2gH}$ Where $K_u =$ Speed Ratio (0.6 to 0.9)

$V_{f1} = K_f\sqrt{2gH}$ Where $K_f =$ Flow ratio (0.15 to 0.3)

Discharge radial $V_{w2} = 0$

$$\eta_h = \frac{W.D/sec}{W/P} = \frac{1}{gH}(V_{w1}u_1)$$

Diameter ratio

$$D_2 = \frac{1}{2}D_1 \ , \ B_2 = 2B_1 \ , \ B_1 = nD_1$$

Breadth ratio $= \dfrac{B}{D}$

$A = K_t \pi D B$ where $K_t =$ Vane thickness factor approx. 0.95

Hydraulic efficiency:

$$\eta_h = \frac{W.D/\sec}{WP} = \frac{R.P.}{W.P}$$

$$\eta_h = \frac{\frac{wQ}{g}(V_{w1}u_1 + V_{w2}u_2)}{wQH}$$

$$\eta_h = \frac{V_{w1}u_1 + V_{w2}u_2}{gH}$$

For max. o/p, $V_{w2} = 0$

$$\eta_h = \frac{V_{w1}u_1}{gH} \text{ (Approximately 85 to 95\%)}$$

Mechanical efficiency

$$\eta_{mech} = \frac{S.P.}{R.P.}$$

Overall efficiency

$$\eta_{overall} = \frac{S.P.}{W.P.} = \eta_h \eta_{mech} \text{ (80 to 90\%)}$$

Working Proportion of Francis turbine:

1) Ratio of width to diameter:

$$n = \frac{B_1}{D_1}$$ Where, B_1, D_1 are width and diameter of wheel at inlet

n ranges 0.10 to 0.45

2) Flow ratio:

$$K_f = \frac{V_{f1}}{\sqrt{2gH}} = \frac{velocity\ of\ flow\ at\ inlet}{therotical\ jet\ velocity}$$

K_f varies from 0.15 to 0.30

3) Speed ratio (K_u)

$$K_u = \frac{u}{\sqrt{2gH}} = \frac{peripheral\ speed\ at\ inlet}{therotical\ jet\ velocity}$$

K_u varies from 0.6 to 0.9

Design of Francis turbine runner:

It involves the determination of size and vane angles. It is designed to develop a known power at a known speed (N) and under known head (H).

Steps:

1) Assume suitable value η_o, η_h, n, K_f, K_t

2) Find discharge from,

$$\eta_{overall} = \frac{S.P}{\frac{W}{P}} = \frac{P}{wQH}$$

$$Q = \frac{P}{\eta_o wH}$$

3) From Q and A , find velocity of flow.

Let, B_1, D_1, t_1 are width, diameter and thickness of wheel at inlet

Z_t = no. of vanes in the runner

Area of flow at inlet, A = $(\pi D_1 - Z_{t1})B_1$

$$= K_{t1}\pi D_1 B_1$$

K_{t1} = vane thickness factor

$$Q = AV_{f1} = K_{t1}\pi D_1 B_1 V_{f1}$$

$$V_{f1} = \frac{Q}{K_{t1}\pi D_1 B_1}$$

Now, $n = \frac{B_1}{D_1}$, $K_f = \frac{V_{f1}}{\sqrt{2gH}}$

$$V_{f1} = \frac{Q}{K_{t1}\pi D_1^2 n} = K_f\sqrt{2gH}$$

$$D_1^2 = \frac{Q}{K_{t1}\pi n K_f\sqrt{2gH}}$$

$$D_1 = \sqrt{\frac{Q}{(K_f\sqrt{2gH})K_{t1}\pi n}}$$

$B_1 = n\,D_1$

4) Tangential velocity (rim velocity)

$$u_1 = \frac{\pi D_1 n}{60}$$

5) Velocity of whirl

$$\eta_h = \frac{V_{w1}u_1}{gH}$$

$$V_{w1} = \frac{\eta_h gH}{u_1}$$

6) Guide vane angle (α)

$$\tan\alpha = \frac{V_{f1}}{V_{w1}}$$

$$\tan\theta = \frac{V_{f1}}{V_{w1} - u_1}$$

Assume, $D_2 = \frac{D_1}{2}$ and $u_2 = \frac{u_1}{2}$

7) Velocity of flow at exit (V_{f2})

$$Q = K_{t1}\pi D_1 B_1 V_{f1} = K_{t2}\pi D_2 B_2 V_{f2}$$

$$\frac{V_{f1}}{V_{f2}} = \frac{K_{t2}\pi D_2 B_2}{K_{t1}\pi D_1 B_1}$$

8) Usually assume, $V_{f1} = V_{f2}$ and $K_{t1} = K_{t2}$

   It gives, $B_2 = 2B_1$

9) Runner vane angle at outlet

   Assume discharge at exit to be radial ($\beta = 90^0$) i.e. $V_{w2} = 0$

$$\tan\phi = \frac{V_{f2}}{u_2}$$

10) No. of runner vanes varies from 16 to 24

    To avoid periodic impulse, no. of runner vanes one or more or one less than no. of guide vanes.

Comparison between Impulse and Reaction turbine:

Sr. no.	Impulse Turbine	Reaction Turbine
1	Works on principle of Impulse momentum principle.	Works on law of angular momentum
2	All available head is converted into K.E. in nozzles	Part of available energy is converted into K.E.
3	Pressure at inlet and outlet of turbine are same.	Pressure at inlet is more compared to an outlet.
4	Runner is not full of water	Runner is with full of water around it.
5	Draft tube is not required	Draft tube is required
6	Possible to regulate flow of water without loss	Not Possible to regulate flow of water without loss
7	Suitable for high heads	Suitable for low or medium heads

Heads for reaction turbine:

1) Gross head (H_g): Difference between head race level and tail race level
2) Net Head (H) : also called as work head or operational head

$$H = \left[\frac{P}{w} + \frac{V^2}{2g} + z\right]_{\text{penstock}} - \left[\frac{P}{w} + \frac{V^2}{2g} + z\right]_{\text{draft tube}}$$

If the draft tube exit is at tail race level and it is assumed as datum with velocity (V_d) as discharge,

$$H = \left[\frac{P}{w} + \frac{V^2}{2g} + z\right]_{\text{penstock}} - \frac{V_d^2}{2g}$$ Where V_d^2=Velocity at exit of draft tube

Axial flow reaction turbines:

If water flows parallel to the axis of the rotation of the shaft, the turbine is known as axial flow turbine. When the head at inlet of turbine is the sum of P.E. and K.E. and during the flow through a part of P.E. is converted into K.E. the turbine known as reaction turbine.

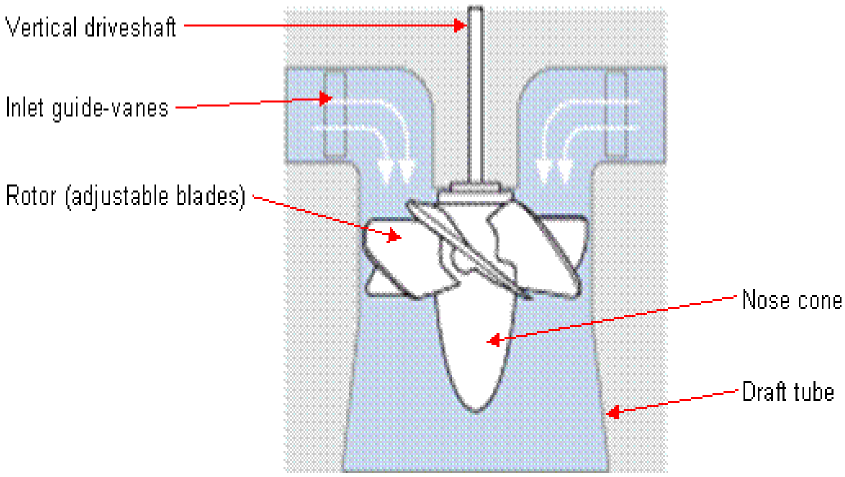
The shaft of an axial flow reaction turbine is vertical. The lower end of shaft is larger known as hub or boss. The vanes are fixed on the hub, it acts as runner.

1) **Propeller turbine:**

   It is used for heads between 4m to 80 m. It provides largest possible flow area. It consists of axial flow runner with 4 to 6 blades of air foil shapes. These are fixed and nonadjustable. It is suitable when load on turbine remain constant.

2) **Kaplan Turbine:**

   At part loads efficiency of propeller turbine is very low since blades are fixed, water runner with shock and eddies are form which reduces efficiency. This defect is removed in Kaplan turbine. The Kaplan blades are adjustable made up of stainless steel. It is compact in size, high speed.



Working Proportions:

$$n = \frac{D_b}{D_o}$$

D_o = outer diameter of runner

D_b = Dia of hub or boss

Area of flow at inlet A = area of flow x velocity of flow

$$= \frac{\pi}{4}(D_o^2 - D_b^2)$$

Discharge, Q = area of flow x velocity of flow

$$= \frac{\pi}{4}(D_o^2 - D_b^2)\ V_f$$

Flow velocity, $V_f = K_f \sqrt{2gH}$

K_f = flow ratio =0.7

$V_{f1} = V_{f2}$

$u = u_1 = u_2 = \frac{\pi DN}{60}$

$u = K_u \sqrt{2gH}$

K_u = speed ratio

Draft Tube:

A pipe of gradually increasing cross-section connected between turbine exit and tail race is called draft tube. Draft tube must be submerged below tail race level.

Purpose:

To have suction head or negative head created at exit of turbine so that installation of turbine above tail race level.

Acts as a recuperator of pressure energy i.e. kinetic energy is converted into pressure energy.

Theory:

V_2= velocity of water at exit of runner or at inlet of draft tube

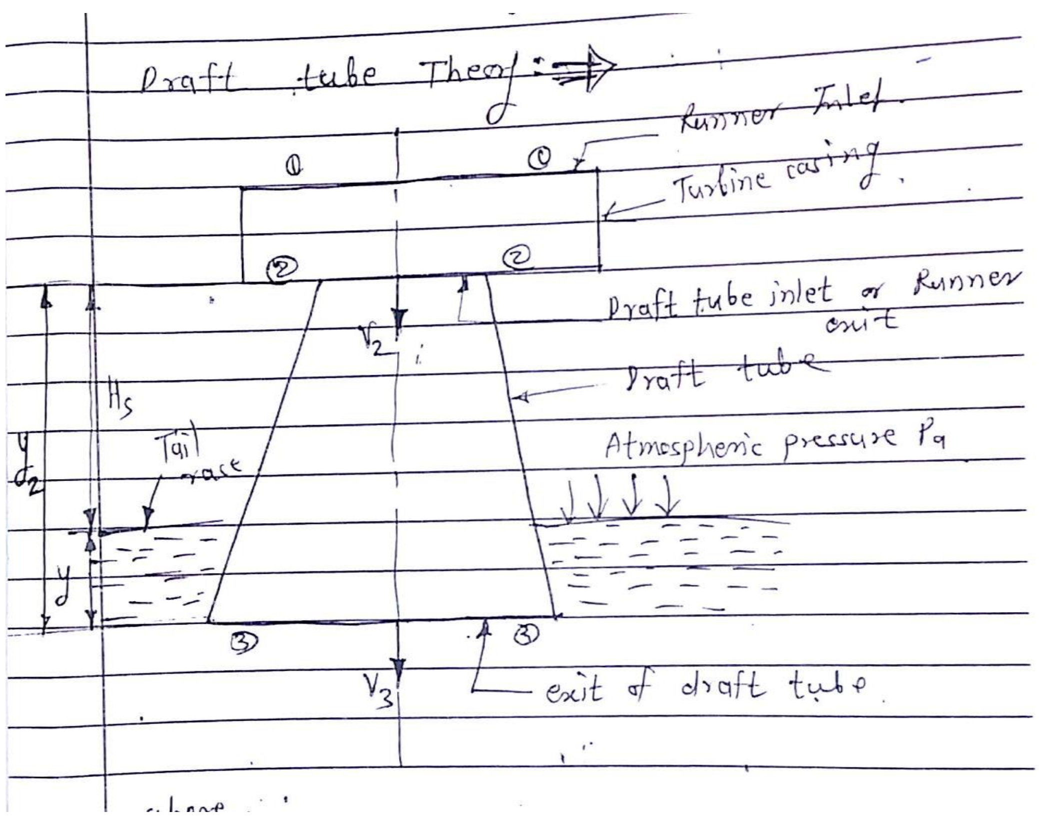
P_a= atmospheric pressure at tail race level

Y= depth of draft tube below tail race level

H_f= hydraulic energy loss in draft tube due to friction

H_s=static suction head

Y_2 = Total height of draft tube



Applying Bernoulli's equation between section (2-2) and section (3-3)

$$\frac{P_2}{w}+\frac{V_2^2}{2g}+y_2=\frac{P_3}{w}+\frac{V_3^2}{2g}+0\ +h_f$$

i.e. $$\frac{P_2}{w}=\frac{P_3}{w}-y_2-\frac{(V_2^2-V_3^2\ )}{2g}+h_f \qquad 1$$

But, $$\frac{P_3}{w}=\frac{P_a}{w}+y \qquad 2$$

Putting equation 2 in 1

$$\frac{P_2}{w}=\frac{P_a}{w}+y-y_2-\frac{(V_2^2-V_3^2\ )}{2g}+h_f$$

But $y - y_2 = H_s$

i.e. $$\frac{P_2}{w}=\frac{P_a}{w}-[H_s+\frac{(V_2^2-V_3^2\ )}{2g}-h_f\ ]$$

This equation shows that pressure at exit of runner drops below atmospheric pressure.

$$\frac{(V_2^2-V_3^2\ )}{2g}= \text{ Dynamic suction head}$$

Efficiency of draft Tube:

It is the Ratio of net gain in pressure head to the velocity head at exit of runner or at entry of draft tube.

$$\eta_d = \frac{net\ gain\ in\ pressure\ head}{velocity\ head\ at\ entry\ of\ draft\ tube}$$

$$\eta_d=\frac{[\frac{V_2^2-V_3^2}{2g}-h_f\ ]}{\frac{V_2^2}{2g}} \qquad 1$$

But friction loss H_f is,

$$h_f=k\frac{V_2^2-V_3^2}{2g} \qquad 2$$

Putting equation 2 in 1

$$\eta_d = \frac{\left[\frac{V_2^2 - V_3^2}{2g} - k\frac{V_2^2 - V_3^2}{2g}\right]}{\frac{V_2^2}{2g}}$$

$$\eta_d = (1-k)\frac{V_2^2 - V_3^2}{V_2^2}$$

Specific Speed:

The speed of turbine which is identical in shape, geometrical dimensions, blade angles, gate opening, etc. which would develop unit power when working under unit head is called as specific speed.

$$\eta_o = \frac{P}{wQH}$$

P = $\eta_o wQH$

P α HQ 1

As we know that, u α V

V α $\sqrt{H}$

u α $\sqrt{H}$ 2

But, $u = \frac{\pi DN}{60}$

u α DN 3

from equation 2 and 3

$\sqrt{H}$ α DN

D α$\frac{\sqrt{H}}{N}$

Discharge through turbine

Q = A V

A = B D

A α B D (since BαD)

A α D^2

Q α $D^2 \sqrt{H}$

$Q \propto \left(\frac{\sqrt{H}}{N}\right)^2 \sqrt{H}$ 4

$Q \propto \frac{H^{\frac{3}{2}}}{N^2}$ 5

From 1 and 5

$P \propto \frac{H^{\frac{3}{2}}}{N^2}\ H$

$P = k\,\frac{H^{\frac{5}{2}}}{N^2}$ 6

Where k is const. of proportion.

Now, P = 1 Kw, H = 1m, N = N_s

K = N_s

Putting K in equation 6

$P = Ns^2\ \frac{H^{\frac{5}{2}}}{N^2}$

$N_s = \sqrt{\frac{N^2 P}{H^{\frac{5}{2}}}}$

$N_s = \frac{N\sqrt{P}}{H^{\frac{5}{4}}}$

Unit Quantities:

1) Unit Speed:

Speed of turbine working under unit head.

$$u = \frac{\pi DN}{60}$$

$$N = \frac{60\sqrt{2gH}}{\pi D}$$

$$N = K_1\sqrt{H} \qquad 1$$

If H = 1 m then N = N_u

$$N_u = \frac{N}{\sqrt{H}}$$

2) Unit Power (P_u)

Power developed by turbine working under unit head

P = wQH = w AVH

$V = \sqrt{2gH}$

$P = w\ A\ \sqrt{2gH}H$

$P = w\ A\ \sqrt{2g}\ H^{3/2}$

$P = K\ H^{3/2}$

If $H = 1m$, $P = P_u$

$P_u = K$

$P_u = \frac{P}{H3/2}$

3) Unit discharge (Q_u):

The discharge of the turbine working under 1m head.

$Q = AV = A\ \sqrt{2gH}$

$Q = K\ \sqrt{H}$

If $H = 1m$, then $Q = Q_u$

$Q_u = K$

Therefore, $Q_u = \frac{Q}{\sqrt{H}}$

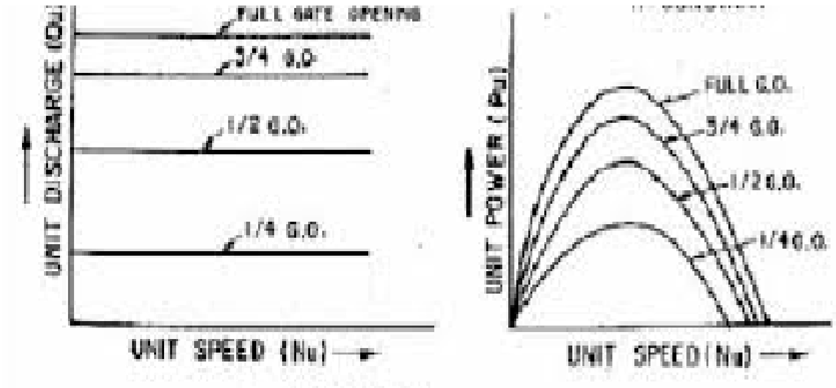
Performance characteristic curves:

The turbine are generally designed to work at particular values of H, Q, P, N and efficiency which are known as desired conditions. But often the turbines are required to work at conditions different from those for which they have been designed; therefore it is essential to determine the exact behavior of turbine under varying conditions by carrying out test either on actual turbine or their small scale model. The result of these is usually graphically represented at resulting curves are known as characteristic curves. These characteristic curves are of the following 3 types;

1. Constant Head Characteristic Curves (main characteristic Curves)
2. Constant Speed characteristic Curves (operating characteristic Curves)
3. Constant efficiency curve

1. **Constant Head Characteristic Curves (main characteristic Curves)**

In order to obtain these curve, the test are performed on the turbine by maintaining a constant head and constant gate opening and the speed is varied by changing the load on turbine. A series of values of N are thus obtain and corresponding to each other value of N, discharge Q at the power output P are measured. From the data of test, the values of Q_u, P_u, N_u and no. are calculated. The curves obtain from pelton wheel and reaction turbines are shown below;

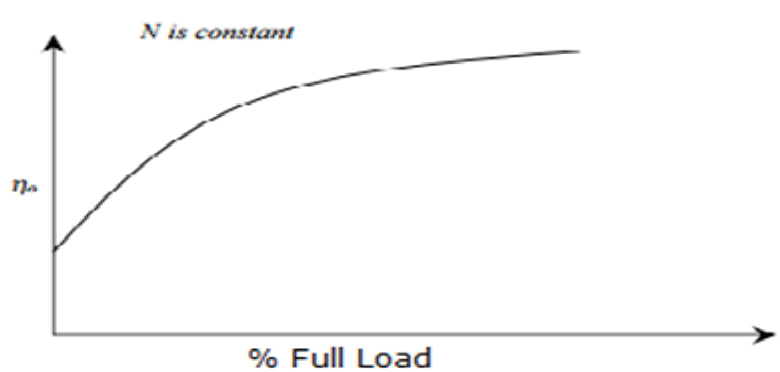


A) For pelton wheel since Q_u depends upon only the gate opening and is independent of N_u, the Q_u vs. N_u plots are horizontal straight line.

B) The curves of P_u vs. N_u and overall efficiency vs. N_u are parabolic in nature for pelton wheel.

2. **Constant Speed characteristic Curves (operating characteristic Curves)**

In order to obtain these curves the test are performed on the turbine by operating them at constant speed. The constant speed is attained by regulating the gate opening there by varying the discharge flowing through turbine as load varies. The load may or may not be constant. The power develop corresponding to each setting of the gate opening is measured and corresponding values of overall efficiency are calculated.



3. Constant efficiency curve